Validation of a building simulation tool using field data for three identical configuration full-serve restaurants using different HVAC Systems

Paul L. Brillhart, PhD
Instructor
University of Illinois, Chicago
Department of Mechanical Engineering
842 West Taylor Street
Chicago, IL 60607-7022

William M. Worek, PhD
Professor
University of Illinois, Chicago
Department of Mechanical Engineering
842 West Taylor Street
Chicago, IL 60607-7022

ABSTRACT

A new building application for a pre-existing HVAC software tool which calculates the benefits of desiccant-assisted HVAC equipment versus the performance of a standard vapor-compression system is used to model the monitored results, see Yborra and Spears (2000), for three full-service restaurants. A standard vapor-compression system, an enthalpy assisted vapor-compression system, and a desiccant-assisted vapor-compression system are compared. The vapor-compression portion of each system is comprised of three rooftop units, specifications for each may be found in Yborra and Spears, "Field-Evaluation of Alternative HVAC Strategies to Meet Ventilation, Comfort, and Humidity Control Criteria at Three Full-Serve Restaurants". The software tool uses DOE 2.1E as a calculation engine which runs in the background. Previously, the software tool could model two different hotel configurations, a quick-serve restaurant, a supermarket, a retail store, an ice arena, a school, a movie theater, a nursing home and a hospital. With the larger eating area, the full-serve restaurant had the capacity for sensible or enthalpy heat recovery from the exhausted air in the sit-down area. Quick-Serve Restaurants (QSR's) were precluded from these energy saving devices as the exhausted air was heavily laden with grease. Still, even with the kitchen exhausts facing away from the rooftop unit (RTU) intakes, the enthalpy wheels showed noticeable loading from grease.

As the field monitoring was performed near Philadelphia, PA, National Renewable Energy Laboratory (NREL) hour-by-hour bin TMY2 meteorological data was used for Philadelphia to model the annual outdoor conditions experienced by each site.

Output was provided in the form of humidity bins, monthly energy usage and cost, as well as total annual gas and electric costs. As the full-serve restaurants were located on the North-Eastern region of the United States, patron comfort was of greater importance to management than annual energy cost savings. Once the model results were determined to properly reflect those of the case studies, the different building equipment types were "moved" around the United States by choosing different bin weather data sets corresponding to Chicago, IL, Atlanta, GA, and Houston, TX. While the default energy rates available in the program are 4 years old, the economic results provide a sound cost comparison.

INTRODUCTION

The HVAC modeling software used here contains revisions to the initial version which will be implemented for the new release. Included in the list of changes are new building types, previously not available. As field data is available for multiple HVAC configurations for a full-serve restaurant, this investigation seeks to confirm the ability of the newest building model to accurately track annual gas and electric energy usage. While energy considerations are a concern to owners and managers of these establishments, especially in light of ASHRAE Standard 62-89, which requires 15 cfm per occupant, patron comfort is of more concern. The longer a patron remains in a full-serve restaurant, the more likely he/she will consume additional drinks and order deserts, both of which are high profit margin items. As many full-serve restaurants have bars, the potential for additional income from a comfortable patron is even greater. The name of the game is to get them in the door and to stay as long as possible. However, full-serve restaurants adhere to the older style of humidity control: turn down the thermostat if a customer feels uncomfortable from excessive humidity. Unfortunately this leads to cold and clammy conditions the likes of which are typically experienced when sensible load conditioning HVAC equipment is used to mitigate humidity.

In addition to comfort, Indoor Air Quality, IAQ, is more of an issue today as seen in ASHRAE 62-89. Also, full-serve restaurants are making a renewed effort to draw in families as opposed to just adults. In most states, managers are faced with the problem of maintaining a comfortable indoor climate one the one hand, and providing adequate IAQ in a facility which includes smoking sections. This is especially true for full-serve restaurants which contain bars as smoking is more prevalent and there has been a recent resurgence of the popularity of the cigar.

SITE SYSTEM SETTINGS

The current configuration of the HVAC modeling software did not allow for altering airstream ventilation control on the front panels, as was required to properly simulate each HVAC system application. Therefore, alterations to the full-serve building specifications through the software's Library folder were performed in accordance to each building's different ventilation design specifications, Yborra and Spears (2000). The Makeup air unit, MAU, as well as the kitchen and restroom exhaust specifications along with the outside air provided directly to the kitchen were accounted for by directly accessing the library files and altering the default building airflow plan. Otherwise, the only inputs specified were those available directly on the front window panels of the HVAC program. While theses parameters were varied in order to accurately simulate field results, the default inputs do provide appropriate settings for comparison of systems across similarly ventilated buildings.

LOCATION, WEATHER, AND ENERGY RATE INPUTS

The Weather and Energy Rate window file was set as Philadelphia, PA for the first set of simulations, see Figure 1. The energy pricing input options did allow for choosing between time of use (TOU) and stepped rate structures along with cutoff type, rate season, demand charges, application of cooling rates and ratchet charges, as well as specific monthly charges, energy cost adjustments and taxes. However, it was decided to use the same pricing schedule as existed at the time of the program's release as separately updating each energy rate for gas and electric schedules for each city modeled was not possible. A reasonable head-to-head comparison of system performance in terms of economics was made with the current rates with the caveat that the

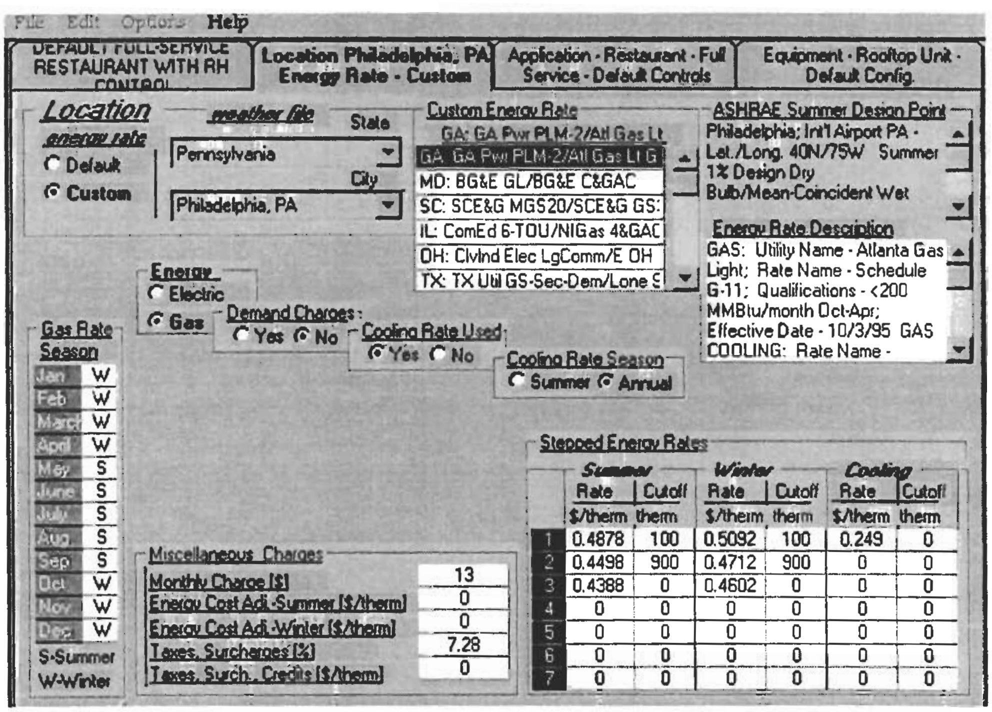

Figure 1 Location, Weather and Energy Rate Window

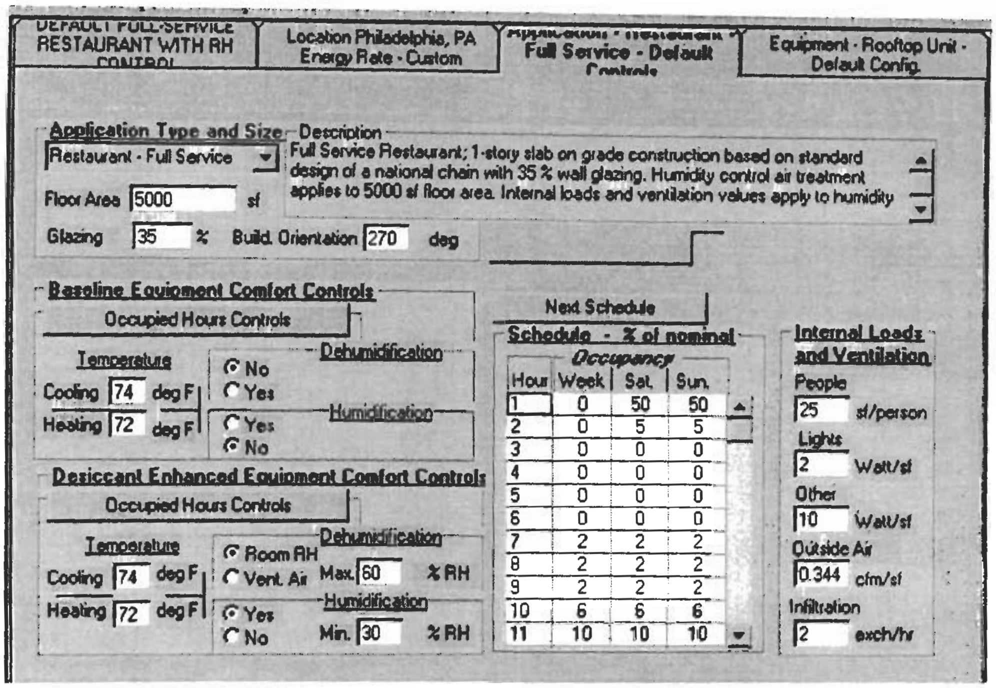

Figure 2 Application Type, Equipment Comfort Controls, Building Schedules, and Load and Ventilation Schedules

vapor-compression system had an advantage in that it was required to process less outside air than the other systems monitored and modeled, as will be explained next.

APPLICATION SETTINGS

In order to accurately model the restaurant, the Application Controls page was modified from its default settings as seen in Figure 2. Specifically, the glazing was changed from 30 to 35% and the building orientation was set at 270 degrees, or west. Also, the Baseline and Desiccant enhanced Equipment comfort controls for occupied and unoccupied hours were altered to reflect those in use, see Table 1. No humidification/dehumidification control was used on the base and enthalpy assisted simulations to reflect the actual control parameters. Additionally, the occupancy, A/C, and Lights/Other schedules were changed to reflect the typical weekly business routines of the sites modeled. Finally, under the Internal Loads section, the amount of ventilation per square foot in the dinning area was 0.344 for the vapor-compression case and 0.634 for the enthalpy and desiccant wheel sites. This was due to local code officials not requiring the standard vapor-compression system site to be in compliance with ASHRAE 62-89 ventilation specifications for outside air. Local officials at the enthalpy and desiccant assisted sites did require the dinning room areas to provide 15 cfm per person to the patrons. The system settings are shown in Table 2.

HVAC EQUIPMENT SPECIFICATIONS

Finally, the Equipment Window, see Figure 3, was setup for controlling indoor conditions to 1% dry-bulb rather than 1% dew-point or both as the sites and most restaurant chains control to dry-bulb only. The desiccant wheel unit used on site was packaged with a post-cooling sensible wheel using outside air and an evaporative cooler. Otherwise the control options in the Equipment Window were not altered from the default settings.

Table 1. Building Comfort Control Set Points

	Standard Electric	Desiccant Enhanced
Cooling Temp./Setback	74/85 (F)	74/85 (F)
Heating Temp./Setback	72/65 (F)	72/65 (F)
Maximum Humidity	----------	60%
Minimum Humidity	----------	30%

Table 2. Building Internal Loads and Ventilation	
Occupancy	24.0 (sf/person)
Lighting	2.0 (Watt/sf)
Other Electric	10.0 (Watt/sf)
Outside Air	1.2 (cfm/sf)
Infiltration	0.10 (exchanges/hr)
Ventilation (vapor-compression)	8.6 (cfm/person)
Ventilation (enthalpy & desiccant)	15 (cfm/person)

RESULTS

The results of the HVAC software tool for the full-serve restaurant are in the form of a comparison of a standard, or augmented, vapor-compression rooftop unit to a desiccant enhanced system. Technical output is available in the form of a sort system size and annual energy consumption summary and a more detailed monthly report which also includes total HVAC load separated into latent and sensible loads for each system simulated. Graphics comparing each system's relative humidity control, monthly gas and electric demand, use, cost,g and annual energy cost are also provided as output. As the total supply and return air to the restaurants was different for each building, see Yborra and Spears (2000), separate simulation runs had to be performed for each equipment type. The output data was used to create Tables 3 – 6, which compare total design capacity, annual gas and electric usage, energy costs, and occupied hours above 60% relative humidity between the standard vapor-compression system, the enthalpy-enhanced system, and the desiccant enhanced system.

FIELD SITE RESULTS

Table 3 compares the simulation results for the standard vapor-compression system to those of the enthalpy-assisted and desiccant-assisted systems. Compared to the systems used in the field, the total design cooling capacity (RT) is 20% higher. However, total electric usage is within 10%. The total vapor-compression tonnage for the enthalpy-assisted system used in the field was the same as the standard vapor-compression system. The simulation results suggest that it may be considerably lowered. Relative to the standard vapor-compression system, the enthalpy-assisted unit exhibited lower energy use, corresponding lower energy costs and better humidity control. As it is a passive energy device, the enthalpy-assisted system was not able to mitigate more than 47 hours of the 1,631 hours above 60%

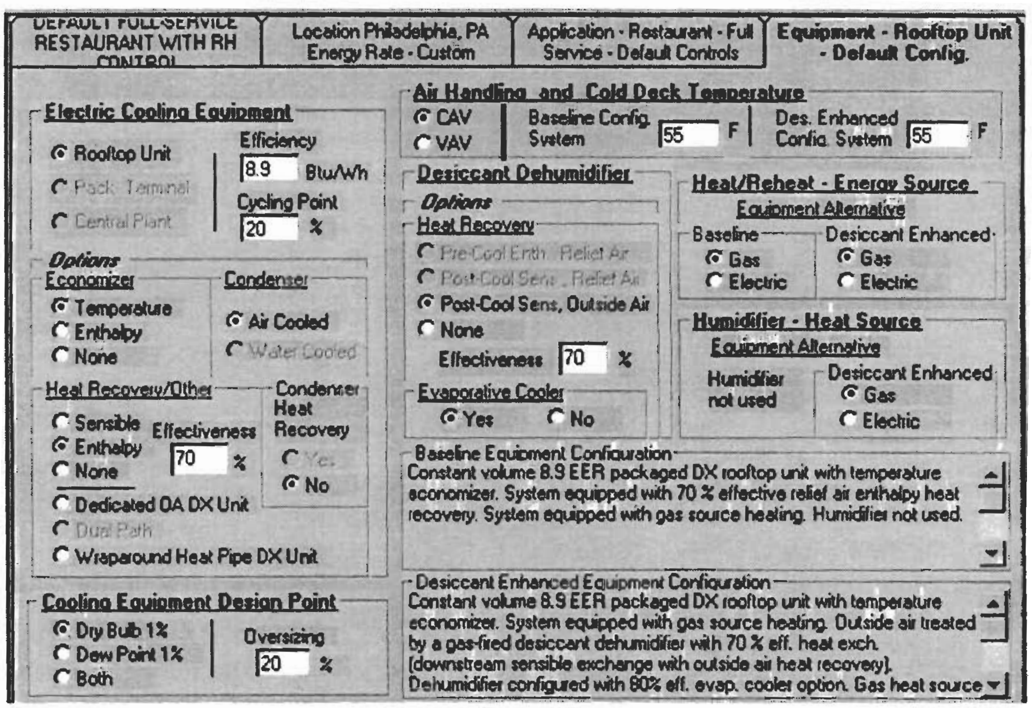

Figure 3 Standard and Desiccant System Equipment Options

Table 3. Equipment Sizing and Energy Use and Costs			
Philadelphia, PA	Standard System	Enthalpy Assisted	Desiccant Assisted
Design Cool. Capacity (RT)	42.2	31.93	27.24
Design Heat.Capacity (kBtu/hr)	1,037	887	1,096
Supply Fan Capacity (CFM)	16,333	13,234	12,948
Annual Elec. Energy Use (kWh)	477,928	461,145	461,104
Annual Gas Energy Use (MBtu)	2,485	2,314	2,873
Annual Electric Energy Cost ($)	35,304	33,352	33,394
Annual Gas Energy Cost ($)	12,490	11,648	13,937
Total Annual Energy Cost ($)	47,794	45,000	47,331
Occupied Hours @ RH > 60%	1,631	1,584	42

relative humidity for the vapor-compression system. It must be noted that the amount of outside air was increased from 8.6 cfm per person for the standard system to 15 cfm per person for the enthalpy-assisted and desiccant-assisted systems.

Also, comparing the enthalpy-assisted system to the desiccant enhanced system results in Table 3, the desiccant enhanced unit had a rooftop tonnage reduction, 31.93 versus 27.24, respectively. The difference in total electric energy usage was negligible, and the desiccant system had an overall increase in gas consumption, 2,873 versus 2,314 for the enthalpy-assisted system. The net result was that the annual energy costs for the desiccant enhanced system were $2,331 higher that the enthalpy–assisted and $463 lower than the standard system.

Offsetting the annual operational cost increase was the deep reduction in desiccant enhanced system operational hours above 60% RH compared to the enthalpy-assisted system, 42 and 1,584 hours, respectively. For a restaurant open 12 hours a day, this equates to 128 less days of operation where customers and staff are outside the comfort zone. While the values for the desiccant enhanced system are different than those of the case site, Yborra and Spears (2000) pointed out that the wall mounted humidity control sensor was not located in an optimal position.

MULTIPLE GEOGRAPHIC LOCATION MODELING RESULTS

In order to see the performance of the different HVAC systems for different geographic locations, the HVAC simulations were "moved" around the United States by using the TMY-2 weather data and local energy rate structures. In addition to Philadelphia, PA, cities modeled were Chicago, IL, Atlanta, GA, and Houston, TX. In this manner, the simulation covers the east, midwest, southern and south western regions of the country.

Results for the three different HVAC applications in Chicago, IL are summarized in Table 4. In terms of annual operational costs, overall annual operating costs were slightly higher than for

Table 4. Equipment Sizing and Energy Use and Costs			
Chicago, Illinois	Standard System	Enthalpy Assisted	Desiccant Assisted
Design Cool. Capacity (RT)	40.70	30.38	26.21
Design Heat.Capacity (kBtu/hr)	1,337	1,145	1,390
Supply Fan Capacity (CFM)	16,121	13,141	12,868
Annual Elec. Energy Use (kWh)	468,639	454,914	454,529
Annual Gas Energy Use (MBtu)	2,856	2,624	3,170
Annual Electric Energy Cost ($)	35,251	33,408	33,6912
Annual Gas Energy Cost ($)	14,321	13,174	15,551
Total Annual Energy Cost ($)	49,572	46,582	49,243
Occupied Hours @ RH > 60%	1,105	1,098	44 .

Table 5. Equipment Sizing and Energy Use and Costs			
Atlanta, Georgia	Standard System	Enthalpy Assisted	Desiccant Assisted
Design Cool. Capacity (RT)	43.88	32.27	28.04
Design Heat.Capacity (kBtu/hr)	886	754	968
Supply Fan Capacity (CFM)	17,081	13,922	13,892
Annual Elec. Energy Use (kWh)	493,366	472,739	475,452
Annual Gas Energy Use (MBtu)	1,945	1,864	2,392
Annual Electric Energy Cost ($)	36,186	34,070	34,281
Annual Gas Energy Cost ($)	9,830	9,426	11,306
Total Annual Energy Cost ($)	46,016	43,496	45,587
Occupied Hours @ RH > 60%	2,262	2,225	10

Philadelphia, PA. However, this was mainly due to a higher design heat capacity. As before, the desiccant enhanced system's annual operational cost was near that of the vapor-compression system, while the enthalpy-assisted system saved approximately $3000. The additional energy cost for running the desiccant system over the enthalpy wheel was due to increased gas consumption. With similar hour total results as the Philadelphia, PA simulations for operation above 60% RH, the desiccant-enhanced system was the only system to truly mitigate the latent load.

Moving the simulation to Atlanta, GA, the total hours above 60% RH increased to 2,226 for the standard vapor-compression system, as seen in Table 5. Collectively, this accounts for one half of the annual open hours for the restaurant. Table 5 shows that the enthalpy-assisted system managed to decrease this by only 37 hours while the desiccant-enhanced system lowered to total hours above 60% RH to 10. Clearly the desiccant-enhanced system is the superior system for controlling indoor humidity levels. The enthalpy-assisted system had the lowest overall annual operating costs, by $2,500, on average. Overall energy costs were down compared to Philadelphia, PA and Chicago, IL. This was due to the decreased system heating requirements.

Finally, bringing the simulation to Houston, Texas, the full-serve restaurants experienced the highest latent loads of the locations tested. As seen in Table 6, the total hours above 60% RH amounted to 322 of the 365 days of the restaurant operating with uncomfortable psychrometric conditions for patrons and employees. As before, the enthalpy-assisted system did minimally lower the total hours from 3,867 to 3,746. Even the desiccant system was not able to entirely mitigate the latent load, however it did decrease it to 111 hours, or 10 working days. As before, the annual energy costs for the standard and desiccant-enhance vapor-compression systems were on par, $47,145 and $47,116, respectively. Again, the enthalpy-assisted system provided the least expensive overall annual operational costs,

Table 6. Equipment Sizing and Energy Use and Costs			
Houston, Texas	Standard System	Enthalpy Assisted	Desiccant Assisted
Design Cool. Capacity (RT)	49.48	36.93	33.98
Design Heat.Capacity (kBtu/hr)	657	555	696
Supply Fan Capacity (CFM)	18,167	14,960	15,342
Annual Elec. Energy Use (kWh)	544,255	515,567	521,046
Annual Gas Energy Use (MBtu)	1,629	1,595	2,334
Annual Electric Energy Cost ($)	38,873	36,468	36,827
Annual Gas Energy Cost ($)	8,272	8,104	10,289
Total Annual Energy Cost ($)	47,145	44,572	47,116
Occupied Hours @ RH > 60%	3,867	3,746	111

$44,576.

CONCLUSIONS

Comparing results of the simulations for three different HVAC systems to the field data gathered, the results consistently over predicted the total rooftop tonnage for the standard vapor-compression system. However, annual electric energy use was within 10% of the field data reported in Yborra and Spears (2000). Like the field data, the simulations did show that standard and enthalpy-assisted vapor-compression systems did not adequately control indoor humidity levels. According to field data, the desiccant system did a far better job of reducing total hours above 60% than either system. The simulation results showed the desiccant system performing better than the field tests. In discussions with Yborra and Spears, it was determined that this was partially due to poor humidistat positioning resulting in the desiccant system not being turned on at all times needed. The desiccant performance curve used in the simulation tool was not from the same manufacturer as the unit monitored. This may, in part, explain the discrepancies in humidity control. Also, the desiccant system had similar annual operational costs as the standard vapor-compression system as was shown in the field test results. This was due to the desiccant system processing 175% more outside air and reducing the moisture content in the air to a comfortable level for occupants.

For different geographic locations, the simulation results showed increasing RT tonnage and increasing hours over 60% RH for hotter and more humid climates. The results further highlighted the increased cooling gas and electric demand and consumption on an annual basis. It should be noted that all locations simulated had problems with humidity control during operational hours, not just Atlanta, GA and Houston, TX. For the vapor-compression system, the humidity control problems it exhibited relative to the desiccant system were in spite of its processing 57% of the outside air as the desiccant system.

The results from this building simulation tool may be used as an accurate tool for performing preliminary screening of different HVAC system types for application in the field. It must be noted that actual systems will perform different in the field owing to different system performance from the performance characteristics of the modeled systems. Additionally, variations in equipment performance characteristics from manufacturer to manufacturer exist. Also, one drawback of using TMY data is that it is possible to omit extreme weather data for both latent and sensible loads. A choice of system over-sizing or running the risk of occasionally experiencing weather conditions outside of the system's design capabilities must be made.

Finally, the simulations clearly demonstrated the need for active humidity control in order to maintain comfortable indoor psychrometric conditions.

REFERENCES

American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., "1997 ASHRAE Handbook of Fundamentals," Atlanta, GA.

"Field-Evaluation of Alternative HVAC Strategies to meet Ventilation, Comfort, and Humidity Control Criteria at Three Full-Serve Restaurants", by Stephen C. Yborra and John W. Spears. Symposium on Improving Building Systems in Hot & Humid Climates, Austin, Texas, 2000.